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LONGITUDINAL VIBRATION OF PROPULSION SYSTEM ON USS SIMON LAKE (AS-33)

by

Gary P. Antonides

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ACCUSTICS AND VIBRATION LABORATORY
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TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

ABSTRACT

Measurements were made on the propulsion system of the USS SIMON LAKE (AS-33) in February 1965 as part of a program to improve analysis procedures used by the Navy for predicting the longitudinal vibration of shaft propulsion systems. The objectives were to find the axial exciting forces and damping associated with the propulsion system of this ship, as well as to determine how the gear case, turbines, condenser, and machinery foundation affect longitudinal vibration. Alternating thrust in the shaft and longitudinal displacement of the gear case, low-pressure turbine, condenser, and machinery foundation were measured. A resonance was found to exist in the operating range, but it is not considered detrimental. The gear case, turbines, and condenser move essentially as one unit. A mass-elastic system derived from measured data includes a lever effect acting on the foundation mass. The exciting forces are lower than usual, except at or near full power.

ADMINISTRATIVE INFORMATION

The work associated with this report was done as a part of the project authorized in BUSHIPS letter Serial 345-326 of 16 July 1963, and funded under S-F013 11 08, Task 01351.

INTRODUCTION

The alternating thrust generated by a ship propeller in an irregular wake causes the propeller shaft and ship propulsion system to vibrate fore and aft. This occurs primarily at blade frequency and often the first critical speed is unavoidably within the operating speed range of the ship. Magnified by the dynamics of the propulsion system, the alternating thrust is occasionally large enough to cause thrust reversal and pounding of the thrust bearing, excessive wear on gears or couplings, or undesirable vibration of pipes or other parts of the propulsion machinery. Unfortunately, the vibratory behavior of shaft-propulsion systems is hard to predict. One or more of these problems may exist on a ship despite a thorough investigation in the design stage.

As part of a program to improve procedures for predicting the longitudinal vibration of shaft propulsion systems, sea trials were conducted in February 1965 on the USS SIMON LAKE (AS-33). This Polaris

submarine tender was built by Puget Sound Naval Shipyard (PSNS) and commissioned in November 1964. The objective of the sea trials was to obtain data with which to find (1) the axial exciting forces and damping, and (2) the effect of the gear case, turbines, condenser, and machinery foundation on longitudinal vibration of the propulsion system. The uncertainty associated with accounting for these factors is what makes predictions of axial response unreliable.

PSNS made natural frequency calculations for axial shaft vibration on SIMON LAKE and the David Taylor Model Basin made a prediction of the response.*

SHIP CHARACTERISTICS

The ship characteristics are given in Table 1, and the propeller arrangement is shown in Figure 1.

TABLE 1

Ship Characteristics

Length:	
Overall	638 feet
Between perpendiculars (LWL)	620 feet
Breadth	85 feet
Depth (to main deck)	57 feet
Draft (DWL)	24 feet
Normal displacement	22,000 tons
Maximum propeller speed	150 rpm
Maximum shaft horsepower	22,500 hp
Trial conditions:	
Draft:	
Forward	18 feet 0 inches
Aft	23 feet 6 inches
Displacement	18,350 tons

*See letter report of February 1964 by Angelos Zaloumis "Longitudinal Shaft Vibration Study of the AS-33."

FULL-SCALE TRIALS

The instruments used in recording are:

7 CEC-type 4-102A vibratory velocity gages.
8 Kulite-Bytrex Corp., semiconductor strain gages.
1 AVL/David Taylor Model Basin rpm indicator.
1 Ampex FR-1300 tape recorder.
1 Monitoring oscilloscope.
1 Directrite oscillograph.
9 CEC-System-D linear/integrating amplifiers.
2 Shaft-strain telemetering systems.
1 TMB calibration source and switch box.

Velocity gages and strain gages were installed as indicated in Figure 2. A wiring diagram of the instrumentation is given as Figure 3.

All the quantities except longitudinal shaft displacement were measured while the ship was accelerating; decelerating; at full speed with left and right full rudder; conducting a crashback; and at steady speeds of 50, 60, 70, 80, 90, 100, 105, 110, 115, 120, 125, 130, 135, 140, 145, and 150 rpm. Shaft displacement was then substituted for the forward shaft strain gage, and steady-speed runs were made at 50, 80, 100, 113, and 125 rpm.

All runs were made at a water depth greater than six times the draft; 8-ft swells coming from the port beam caused the ship to roll about 10 deg during the trials.

TRIAL RESULTS

Blade frequency thrust and vibration were analyzed for all stations and all speeds. Alternating thrust at double blade frequency was analyzed to determine the second mode frequency. A sample oscillograph record (Figure 4) indicates the predominance of blade frequency and the modulation which is typical of the signal recorded on all channels. Figures 5 through 12 plot the average and peak blade-frequency displacements and alternating thrust against rpm for steady-speed runs. Figure 13 plots the average alternating thrust for double-blade frequency against rpm. These data were obtained by filtering the signal electronically to get blade or double-blade

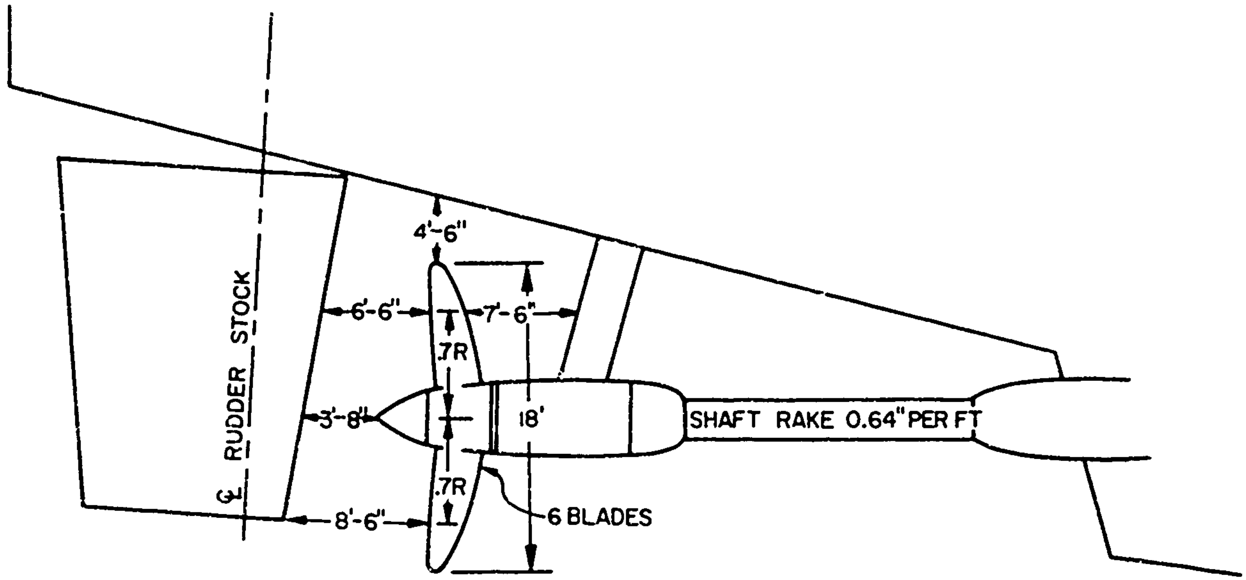


Figure 1 - Propeller Arrangement

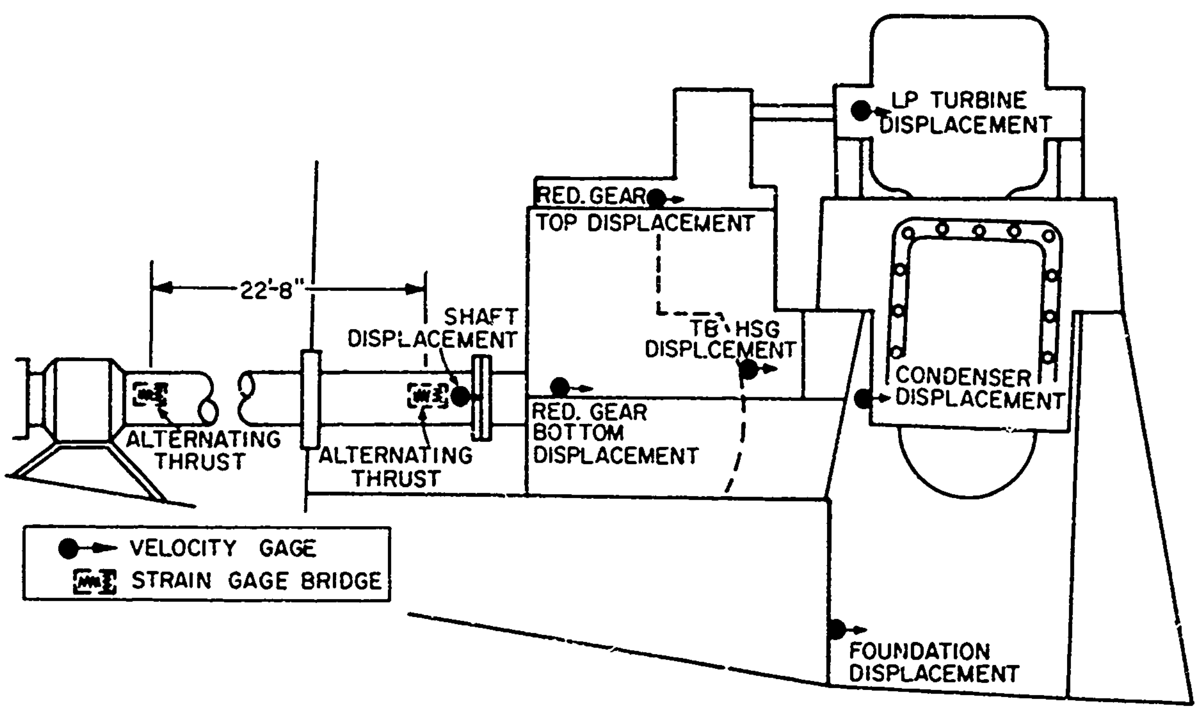


Figure 2 - Main Propulsion Plant of SIMON LAKE Showing Location and Orientation of Gages

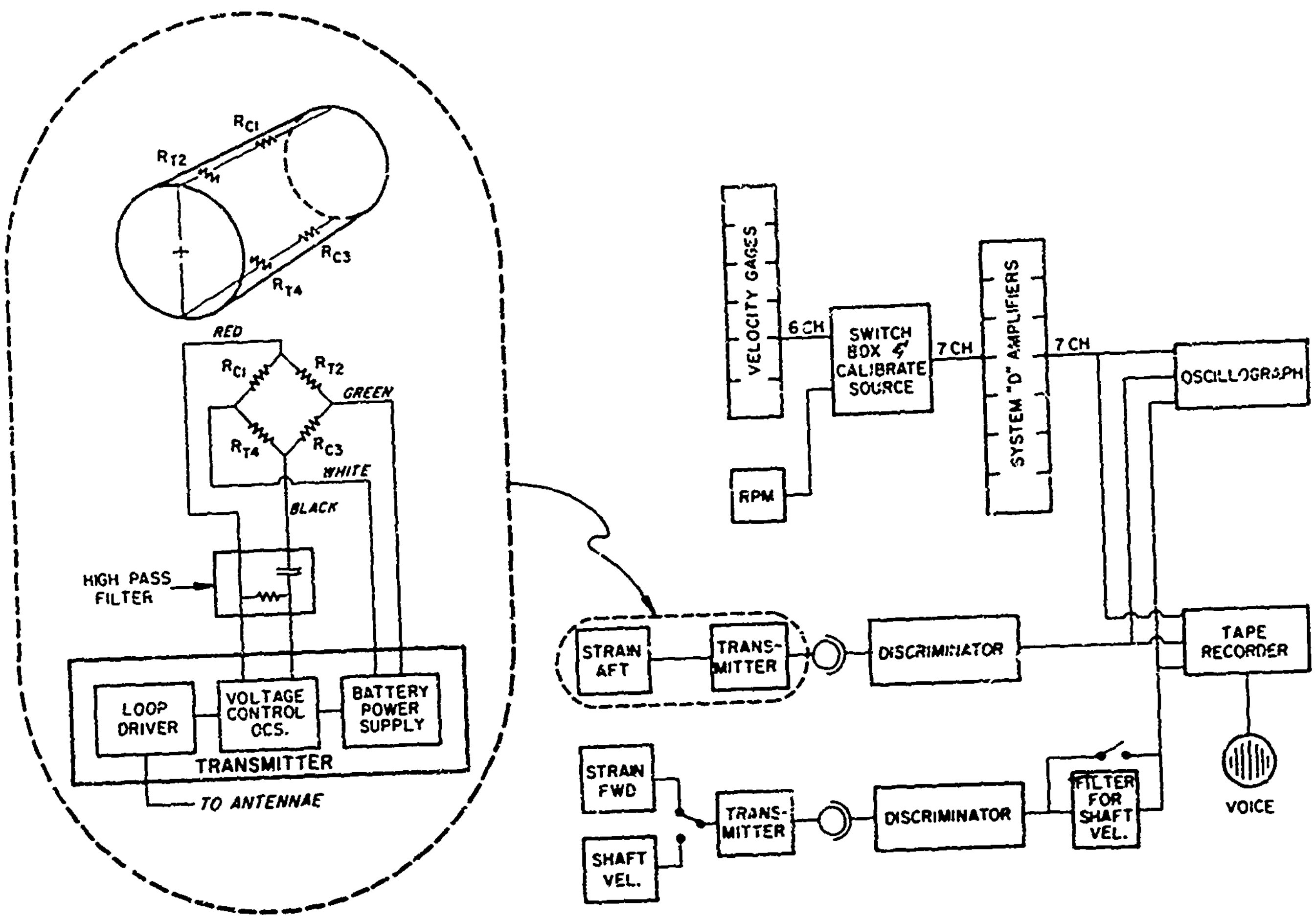


Figure 3 - Wiring Diagram of Instrumentation

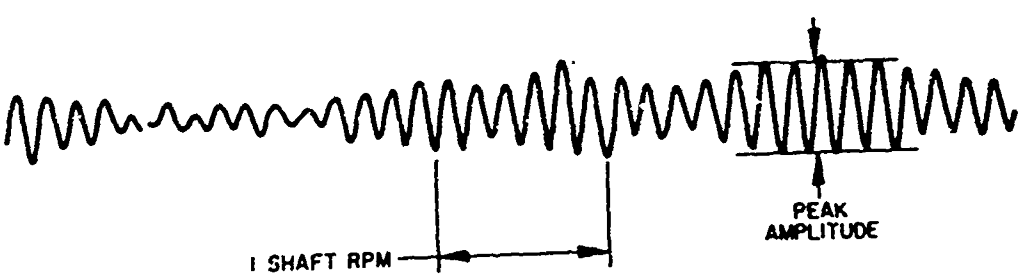


Figure 4 - Sample Oscillograph Record Taken from Thrust-Bearing Housing at 130 RPM

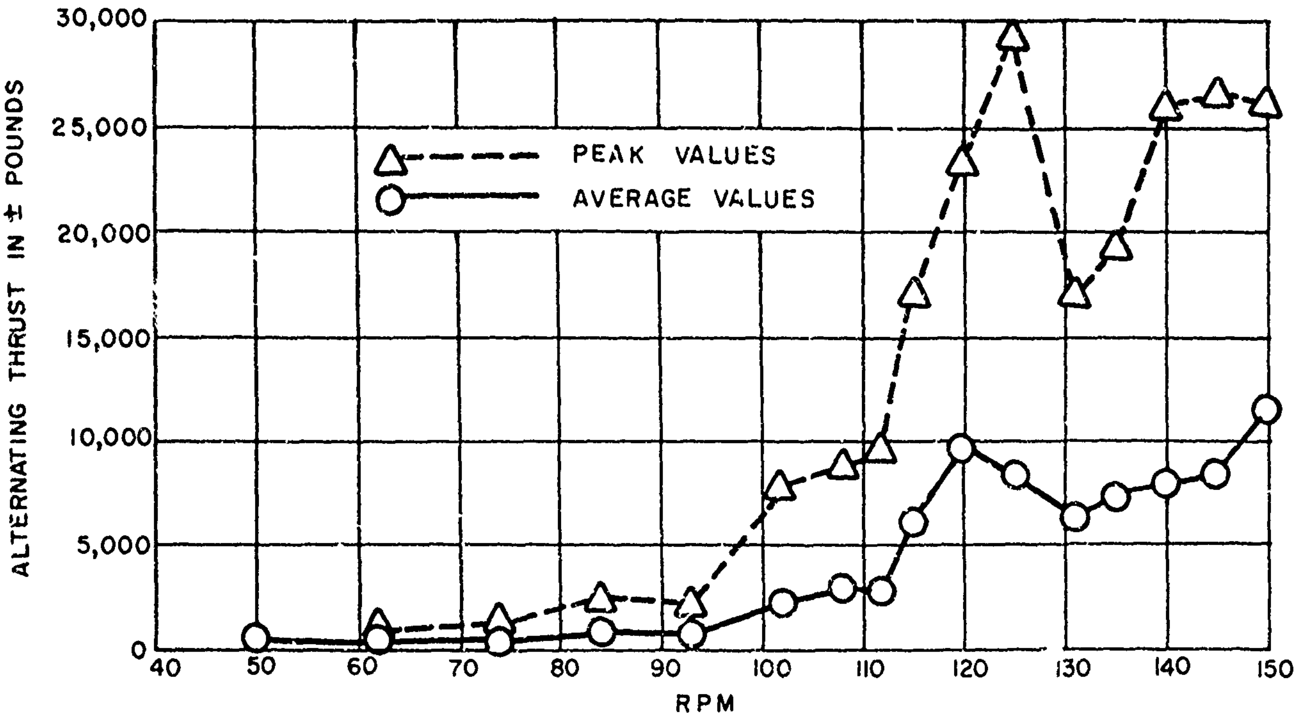


Figure 5 - Blade Frequency Alternating Thrust in Shaft (Forward) versus RPM

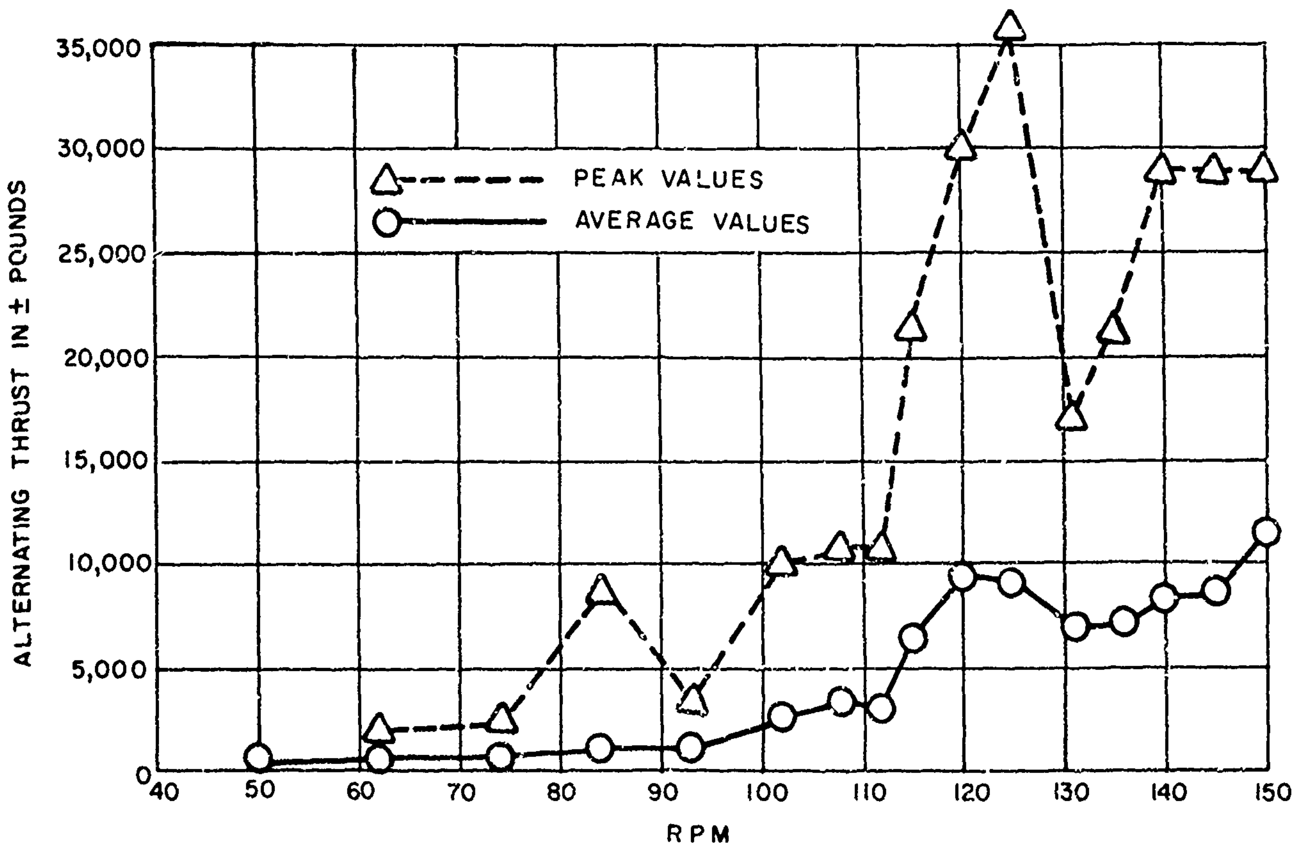


Figure 6 - Blade Frequency Alternating Thrust in Shaft (Aft) versus RPM

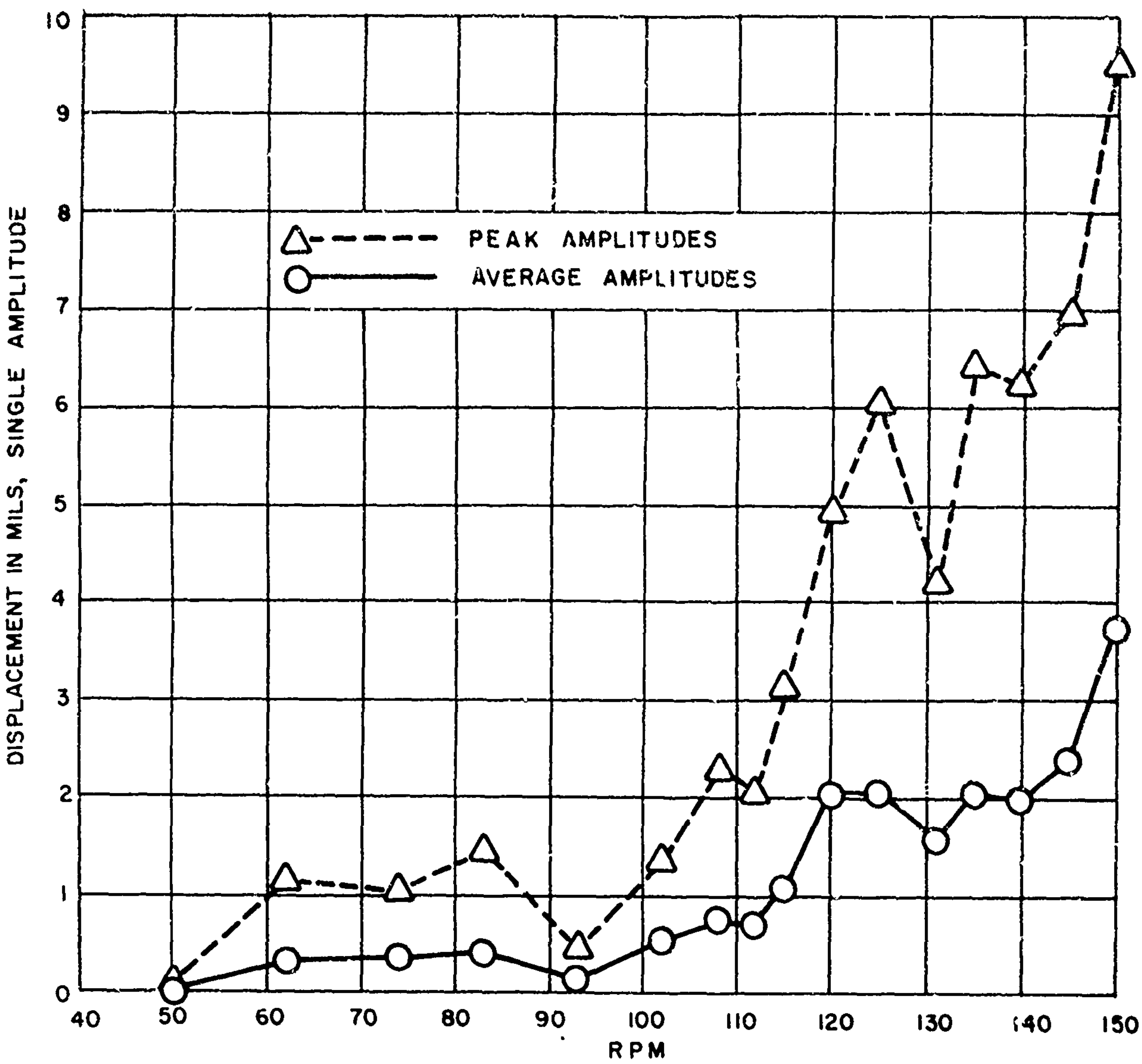


Figure 7 - Longitudinal Blade Frequency Displacement of Gear Case Top versus RPM

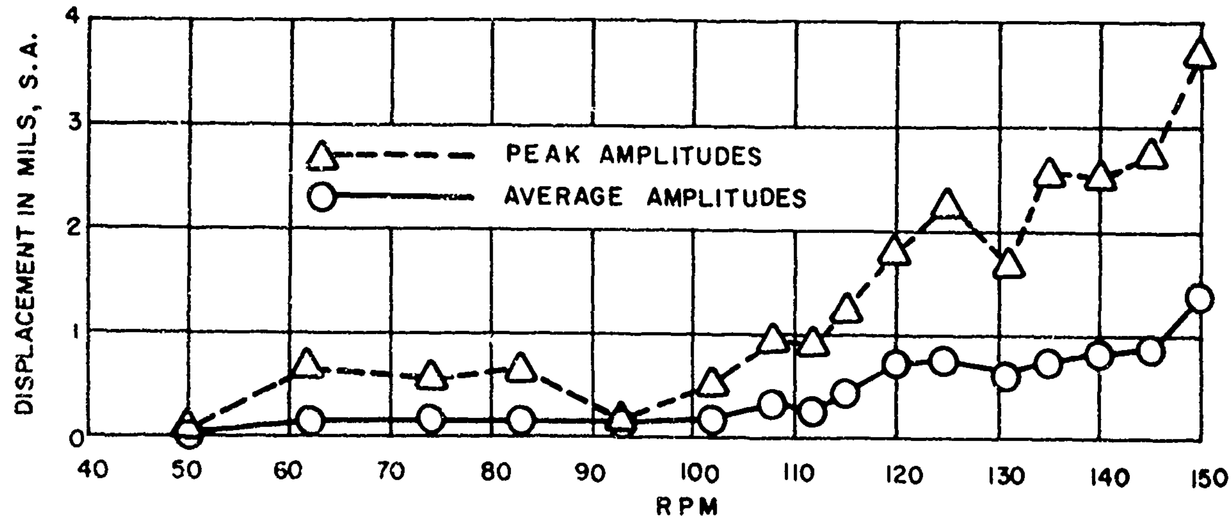


Figure 8 - Longitudinal Blade Frequency Displacement of Gear Case Bottom versus RPM

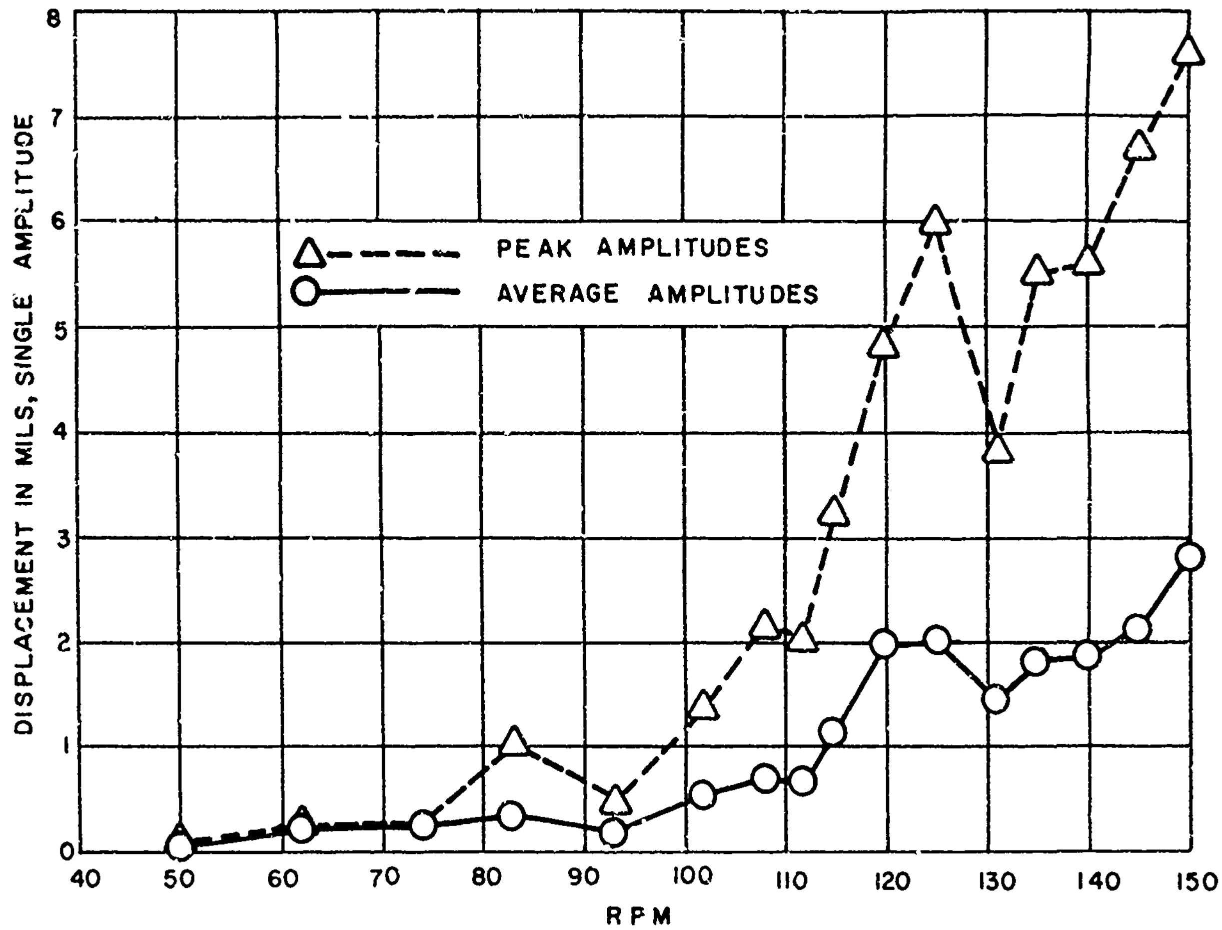


Figure 9 - Longitudinal Blade Frequency Displacement of Thrust Bearing Housing versus RPM

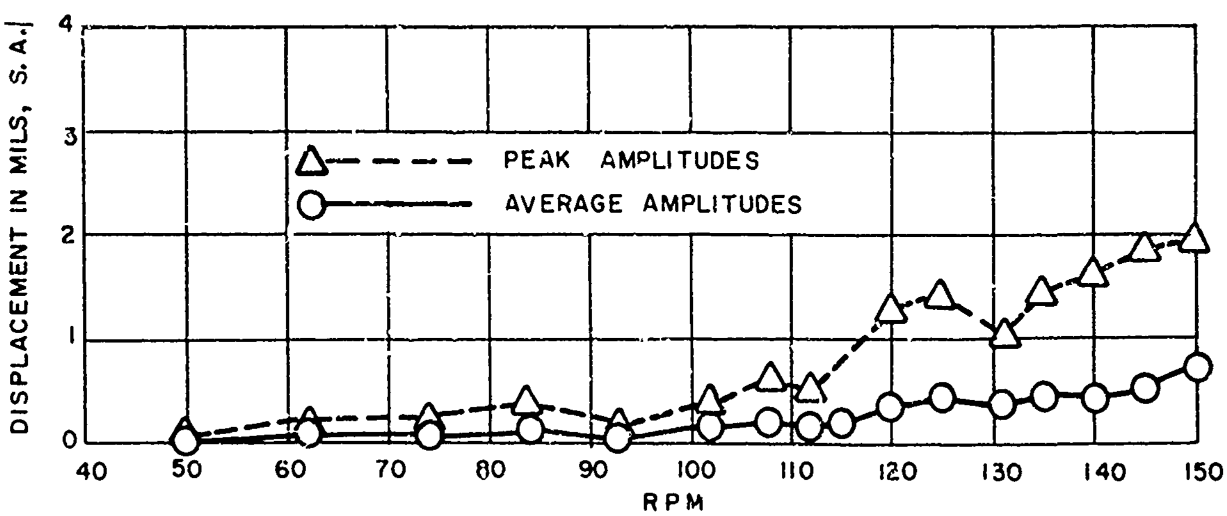


Figure 10 - Longitudinal Blade Frequency Displacement of Foundation versus RPM

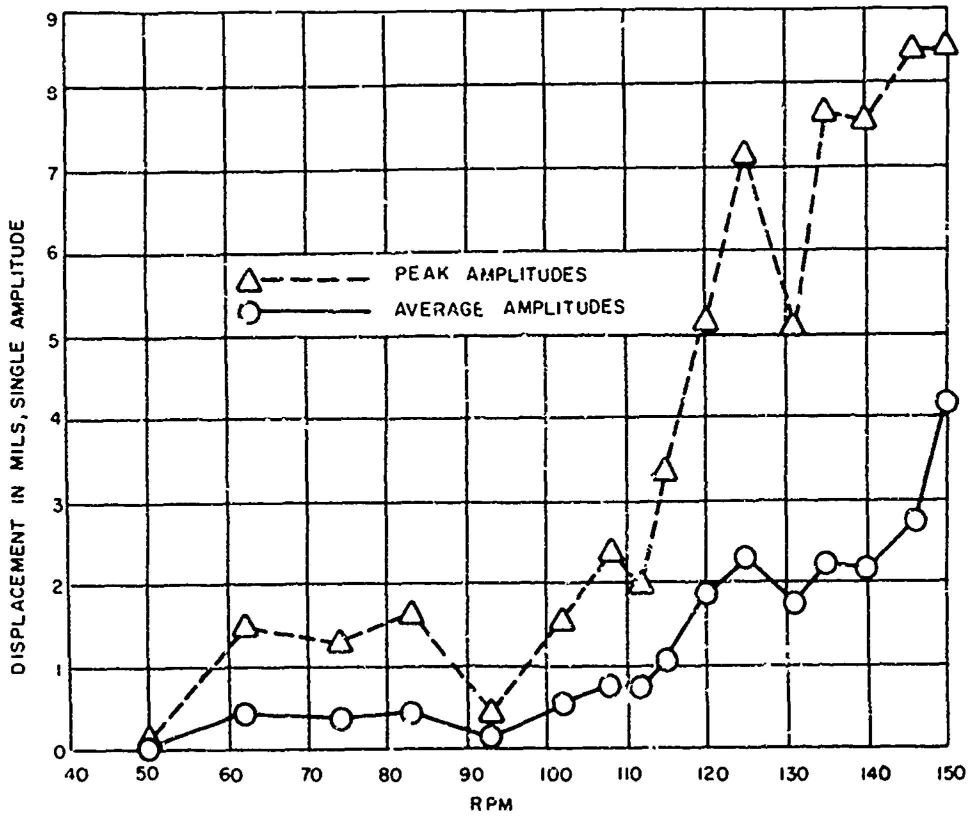


Figure 11 - Longitudinal Blade Frequency Displacement of Low Pressure Turbine versus RPM

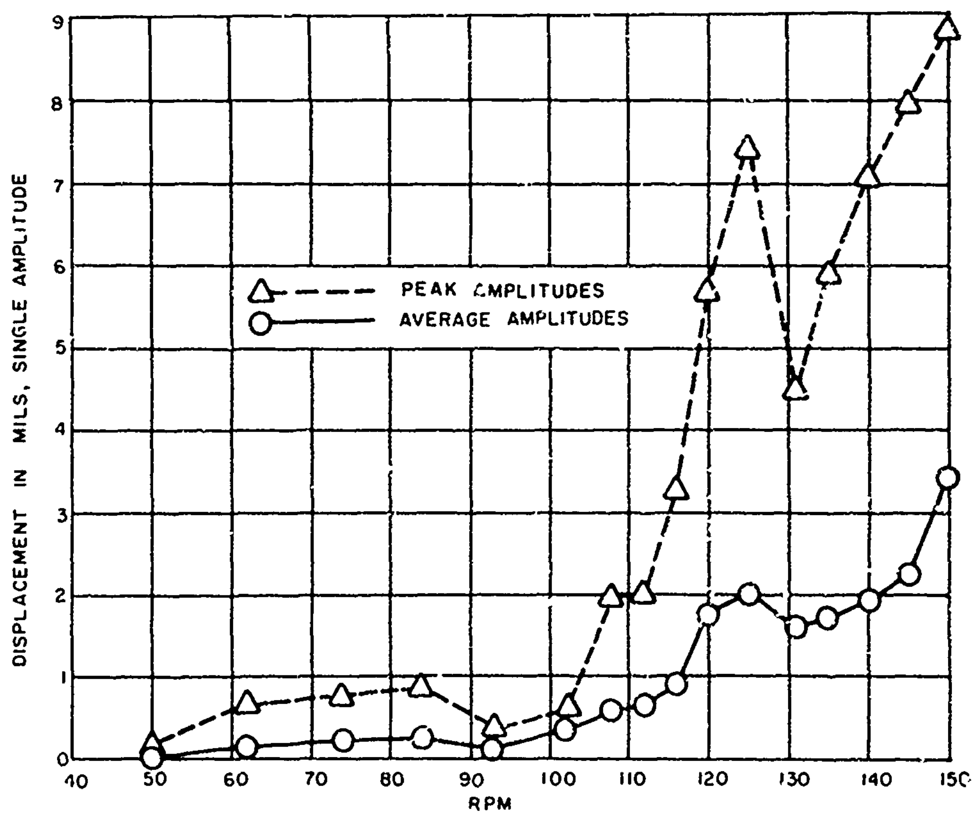


Figure 12 - Longitudinal Blade Frequency Displacement of Condenser versus RPM

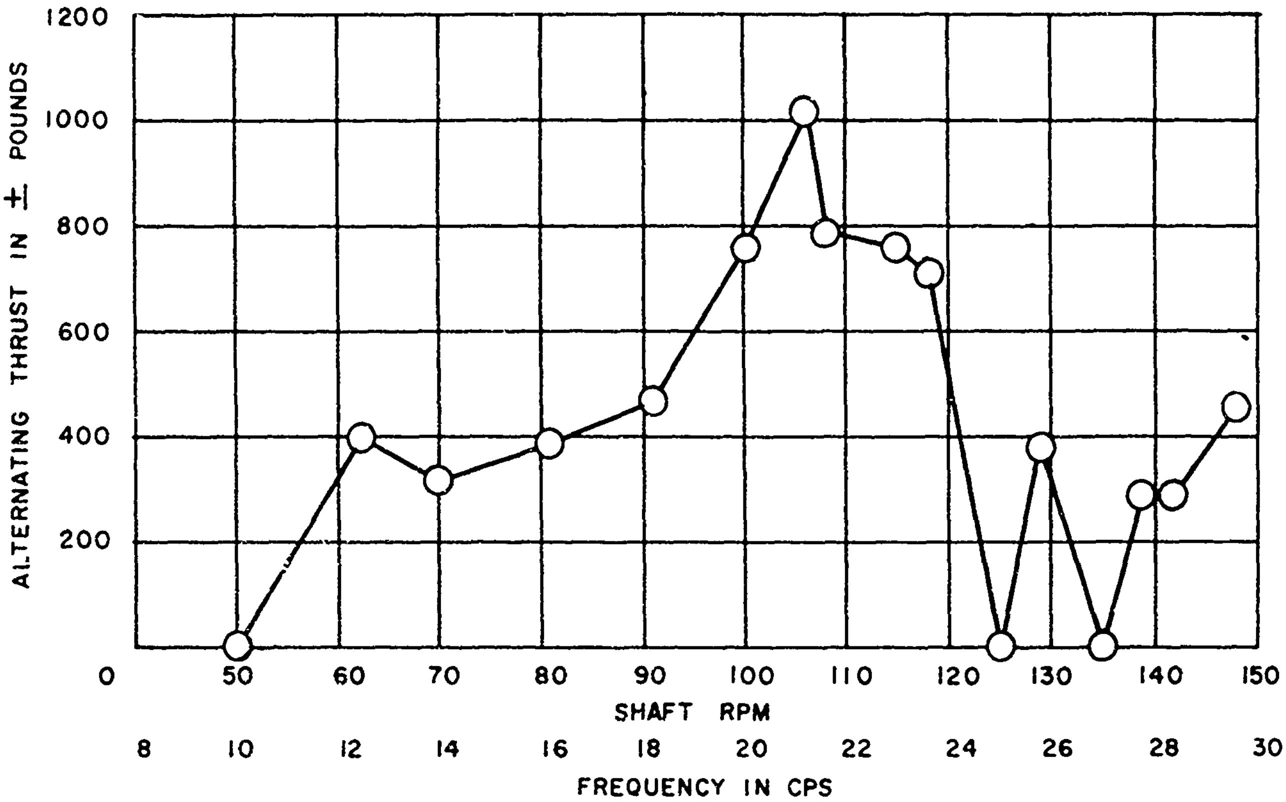


Figure 13 - Average Double Blade Frequency Alternating Thrust in Shaft (Forward) versus RPM

TABLE 2

Maximum Blade Frequency Amplitudes* of Longitudinal Vibration and Thrust during Maneuvers

Station	Maneuver			
	Acceleration (Peak at 140 rpm)	Right turn (at 136 rpm)	Left turn (at 134 rpm)	Crashback (peak at 46 rpm ahead)
Thrust bearing housing	5.3	6.4	8.5	6.4
Reduction gear top	7.3	6.3	10.4	8.3
Reduction gear bottom	2.0	4.0	5.0	5.0
Foundation	1.0	2.1	3.1	2.1
Turbine	6.2	7.3	12.5	10.4
Condenser	6.0	8.0	10.0	8.0
Alternating thrust †	±15,400 lb	±30,900 lb	±38,700 lb	±30,900 lb

*All displacements are expressed in mils, single amplitude. Amplitudes during maneuvers were determined from oscillograph records.

†Full power mean thrust is 240,000 lb.

frequency and then by either electronically averaging the signal or measuring the peak values on each run. The peak values are of primary concern since machinery and structures must be designed for maximum vibratory forces. To determine the shape of the amplitude versus rpm curve, however, the average values may be more suitable. It is possible that occasional large increases in amplitudes due to waves, ship motion, or some other extraneous factor may distort the peak amplitude versus rpm curves to the extent that they give a false indication of the resonant frequency or damping. For this reason, the resonant frequency and damping were derived from the average amplitude versus rpm curves. It should be noted, however, that there were no significant distortions in the shape of the peak amplitude curves obtained from this trial. This is evidenced by a more or less constant factor of three between peak and average values at all stations and at all speeds. If subsequent trials on other ships show the same pattern, it will be possible to rely on peak values for determination of resonant frequencies and damping as well as for design purposes.

All stations were vibrating in phase at all steady speeds. The maximum blade frequency amplitudes for maneuvers are given in Table 2.

The velocity pickup strapped to the shaft seemed to work well despite centrifugal force. The highest shaft speed attained with this gage was 125 rpm. At this speed the maximum displacement of the shaft was about 4 mils, single amplitude, and that of the thrust-bearing housing was 3 mils, single amplitude; both were at blade frequency.

A peak was apparent at all stations at about 122 rpm due to first mode resonance. At this resonance, the measured alternating thrust in the line shaft at blade rate was as high as ± 24 percent of the mean thrust. This percentage is taken from the peak value in Figure 6.

The alternating thrust at double-blade frequency (Figure 13) indicates that the second mode natural frequency is 21 cps.

ANALYSIS

This analysis has been made to define, as nearly as possible from the trial data, an equivalent mass-elastic system, the exciting forces, and damping associated with the SIMON LAKE propulsion system.

MASS-ELASTIC SYSTEM

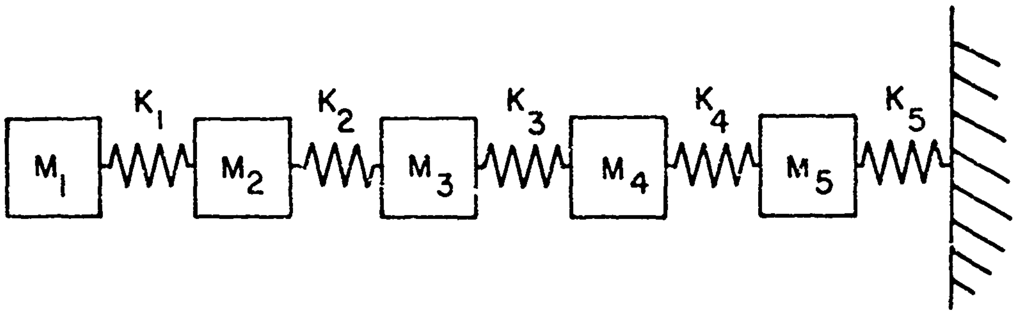
The structure of the propulsion system from the propeller through the thrust bearing is comparatively simple. The values of mass and stiffness associated with this portion of the system can be adequately defined in accordance with established procedures. Therefore, the values of the parameters of the mass-elastic system used in the Model Basin prediction and given in Figure 14 are considered correct. The other two parameters, foundation stiffness and effective foundation mass, are difficult to estimate.

The estimation of foundation stiffness is normally based on the results of experimental measurements on similar installations. Calculations to determine this parameter are complex and have not matched experimentally determined values closely enough to be useful in design prediction.

The effective foundation mass is normally assumed to be the actual mass of the reduction gear case, first reduction gears and pinions, low pressure and high-pressure turbines, and condenser. There are two considerations which may complicate this otherwise straightforward general procedure.

First, the condenser and turbine are mounted on some ships with enough compliance to be considered as a separate mass. On SIMON LAKE, however, the amplitudes of the gear case top, thrust-bearing housing, turbine, and condenser are approximately the same, so that a single mass representation is considered appropriate.

Second, since the greater part of the foundation mass is above shaft level and since it is structurally attached at the bottom, a "lever effect" may tend to increase the amplitudes of most of the foundation mass which is above the shaft level. There is evidence of this in the measurements taken (Figure 15). The level of each gage is projected to a vertical scale, and the displacements are plotted for eight different speeds. The points considered to be the most reliable indicators of the lever motion are indicated with solid dots. The thrust-bearing housing, condenser, and turbine displacements, may be unreliable for determining lever motion, because (1) local deformation of the thrust-bearing housing



Parameter	Description of Parameter	Value of Parameter
M_1	Mass of propeller, including 60 percent for virtual mass, plus 1/2 propeller shaft (lb-sec^2/in)	183
M_2	1/2 mass of prop-shaft, plus 1/2 mass of stern tube shaft	90
M_3	1/2 mass of stern tube shaft, plus 1/2 mass of line shaft	70
M_4	1/2 mass of line shaft, plus mass of Bull gear and second reduction pinions	128
M_5	Effective mass of condenser, turbines, gear case, first reduction gears, and pinions	unknown
K_1	Stiffness of prop. shaft (lb/in)	18.1×10^6
K_2	Stiffness of stern tube shaft	11.1×10^6
K_3	Stiffness of line shaft	14.7×10^6
K_4	Thrust bearing stiffness	6.0×10^6
K_5	Foundation stiffness	unknown

Figure 14 - Mass-Elastic System of SIMON LAKE Propulsion Plant

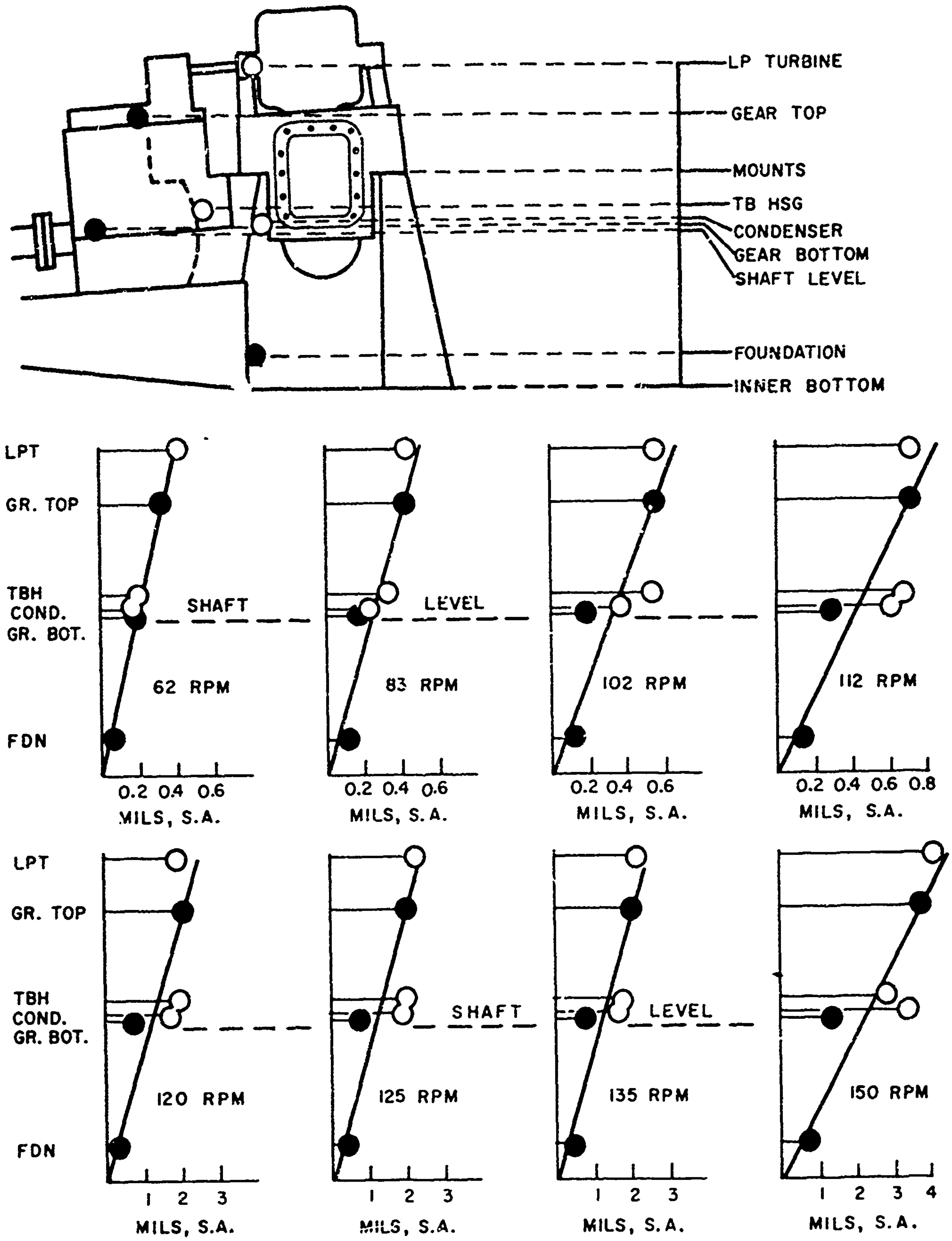


Figure 15 - Profile of Propulsion System Vibration Showing Lever Effect

makes that displacement larger than other parts of the gear case at the same level and (2) since the turbine and condenser are supported only at one point on the gear case, they probably move longitudinally but do not rotate about an athwartship axis.

For calculations, it seems reasonable to assume that the inner bottom of the ship does not vibrate longitudinally and that the longitudinal vibration of any part of the foundation mass is proportional to the height above this point of rotation, as indicated by the straight lines of Figure 15. To account for the lever effect, the mass-elastic system of Figure 16 might be used. In this simplified representation, the parameters are the same as previously with K_5 and M_5 unknown. The additional parameters are:

a is the height of the shaft ℄ above the ship inner bottom,

b is the "effective height" of the foundation mass above the ship inner bottom.

x_b is the longitudinal displacement of M_5, and

x_a is the longitudinal displacement of the point on the lever at shaft level. The last two quantities are used in place of X_5.

In the equations of motion, the inertial force of M_5 is referred to shaft level and is equated to the applied spring forces. Assuming a small angle of rotation of the lever, the inertial force of M_5 at a height of b is $M_5\ddot{x}_b$. Because of the lever, when referred to shaft level this force is $\left(\frac{b}{a}\right)M_5\ddot{x}_b$. Since $\ddot{x}_b = \left(\frac{b}{a}\right)\ddot{x}_a$, substitution results in an inertial force of $\left(\frac{b}{a}\right)^2 M_5\ \ddot{x}_a$. The only difference between the inertial force of M_5 of the in-line system (Figure 14) and that of the lever system is the factor of $\left(\frac{b}{a}\right)^2$.

To determine the corresponding inertial force in an actual propulsion plant, it is necessary to break down the foundation mass into severl masses m_i, determine the height b_i of each, and sum the inertial forces of each mass to get the total inertial effect of the foundation mass F_{M_5}:

$$F_{M_5} = \sum_i \left(\frac{b_i}{a}\right)^2 m_i\ \ddot{X}_a$$

There are at least two complications to this procedure:

1. The turbine and condenser are supported at only one point on the gear case. Consequently, the lever rotation is probably not effectively transmitted from the gear case to the turbine and condenser. If this is so, b_i for the turbine and condenser should be the height of the mounts.

2. The turbine rotors and the water in the condenser may not move with the rest of the foundation mass.

Because of these and possibly other complications, an empirical approach may be more appropriate. The ratio of the "effective height" of the foundation mass to the shaft height $\frac{b}{a}$ may be nearly the same for all turbine-double reduction gear plants since their arrangement and mass distribution are similar.

The two doubtful parameters $\left(\frac{b}{a}\right)$ and K_5 can be determined from two Holzer tables, one for the fundamental frequency and one for the second natural frequency. The frequencies are known from trial data. The last row of each table is written in terms of $\frac{b}{a}$ and K_5, expressed in equation form, and solved. These computations are shown on page 17. Note that the "effective foundation mass" is the product of the actual mass and $\left(\frac{b}{a}\right)^2$.

The value obtained for K_5 is closer to the PSNS calculated value (20×10^6) than the value used for prediction (10×10^6). It is considered likely that the lever effect described is typical of marine propulsion plants, and that discrepancies between calculated and experimental values of foundation stiffness have been due, in part, to not accounting for this effect.

EXCITING FORCES AND DAMPING

A digital computer was used to obtain amplitude versus rpm curves of the mass-elastic system described above. These curves are compared with the measured response to obtain the exciting forces and damping. The inputs to the computer were the above mass-elastic system with the derived values of (b/a) and k_f and with a single value of damping at the propeller. A damping value C_p of 2760 lb-sec/in. was originally used for prediction. This was obtained by using the Rigby empirical constant of

First Mode: $f_n = 12\ 2$ $\omega_n = 76.7$ $\omega_n^2 = 5870$						
M	$M\omega^2/10^6$	X	$M\omega^2 x/10^6$	$\Sigma M\omega^2 x/10^6$	$K/10^6$	ΔX
183	1.072	1.000	1.072	1.072	18.[illegible]	0.059
90	0.528	0.941	0.497	1.569	[illegible]	0.142
70	0.410	0.799	0.327	1.896	14.[illegible]	0.129
128	0.750	0.670	0.502	2.398	6.0	0.400
$525\left(\frac{b}{a}\right)^2$	$3.070\left(\frac{b}{a}\right)^2$	0 270	$0.830\left(\frac{b}{a}\right)^2$	$2.398 + 0.830\left(\frac{b}{a}\right)^2$	$K_S/10^6$	$\frac{2.398 + 0.830\left(\frac{b}{a}\right)^2}{K_S/10^6}$

Second Mode: $f_n = 21.0$ $\omega_n = 132$ $\omega_n^2 = 17400$						
M	$M\omega^2/10^6$	X	$M\omega^2 x/10^6$	$\Sigma M\omega^2 x/10^6$	$K/10^6$	ΔX
183	3.18	1.000	3.18	3.18	18.1	0.176
90	1.57	0.824	1.30	4.48	11.1	0.403
70	1.22	0.421	0.513	4.993	14.7	0.340
128	2.23	0.081	0.181	5.174	6.0	0.861
$525\left(\frac{b}{a}\right)^2$	$9.13\left(\frac{b}{a}\right)^2$	-0.780	$-7.12\left(\frac{b}{a}\right)^2$	$5.174 - 7.12\left(\frac{b}{a}\right)^2$	$K_S/10^6$	$\frac{5.174 - -7.12\left(\frac{b}{a}\right)^2}{K_S/10^6}$

From the first mode: $$\frac{2.398 + 830\left(\frac{b}{a}\right)^2}{K_S/10^6} = 0.270$$

From the second mode: $$\frac{5.174 - 7.12\left(\frac{b}{a}\right)^2}{K_S/10^6} = -0.780$$

Rearranging and solving: $$0.830\left(\frac{b}{a}\right)^2 - 0.270\ K_S/10^6 = -2.398$$

$$7.12\left(\frac{b}{a}\right)^2 - 0.780\ K_S/10^6 = 5.174$$

$$\left(\frac{b}{a}\right)^2 = \frac{\begin{vmatrix} -2.398 & -0.270 \\ 5.174 & -0.780 \end{vmatrix}}{\begin{vmatrix} 0.830 & -0.270 \\ 7.12 & -0.780 \end{vmatrix}} = 2.54$$

$$K_S/10^6 = \frac{\begin{vmatrix} 0.830 & -2.398 \\ 7.12 & 5.174 \end{vmatrix}}{\begin{vmatrix} 0.830 & -0.270 \\ 7.12 & -0.780 \end{vmatrix}} = 16.6$$

$$\boxed{\frac{b}{a} = 1.6 \quad K_S = 16.6 \times 10^6}$$

16.5 lb-sec/in. per square foot of developed blade area.[1] On the basis of limited data, Rigby also noted that damping seems to increase with the number of blades and suggested that 39.5 lb-sec/in. per foot of blade-edge length might be more appropriate.

Since the measured resonant amplitudes are fairly low, indicating high damping, the latter criteria of Rigby is tentatively used, resulting in a C_p of 4100 lb-sec/in. This is more accurately determined later.

Exciting Forces

To this system (Figure 16) an exciting force of ±1000 lbs is applied at the propeller throughout the frequency range. The resulting response is given in Figure 17.

In the mass-elastic system, K_3 corresponds to the section of shaft where the strain gages were installed. The force in K_3 can be determined by multiplying the stiffness of K_3 by the difference in the displacements of M_3 and M_4. That force, as a function of frequency for a constant ±1000 lb at the propeller, is given as curve (a) in Figure 18. By the principle of superposition, the ratio of actual alternating thrust in the line shaft, which is the measured value, to the computed force in K_3, which is the computer output, is the same as the ratio of actual exciting force at the propeller, which is the unknown, to the computer input, which is ±1000 lb. It is assumed that the mass-elastic system accurately represents the propulsion system. From this relationship, the derived exciting force is found (Figure 18). The hump near resonance in the derived exciting force (curve C) occurs because the value of damping used is slightly inaccurate. The damping is more accurately determined later. Note that average measured values of thrust were used. If peak measured values were used, the derived exciting force would be about three times as large.

Most mass-elastic systems used for longitudinal calculations are good only in the area of the first mode. At full power on SIMON LAKE, the

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[1] References are listed on page 27.

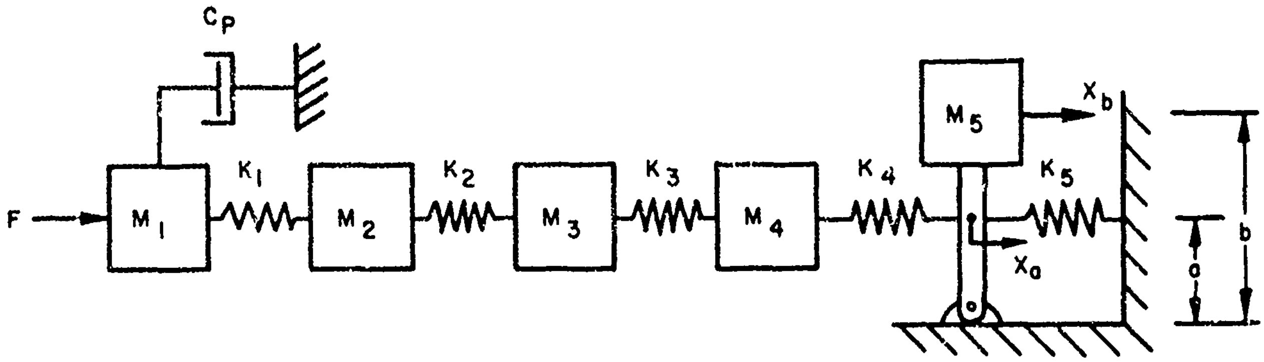

Figure 16 - Mass-Elastic System Accounting for Lever Effect

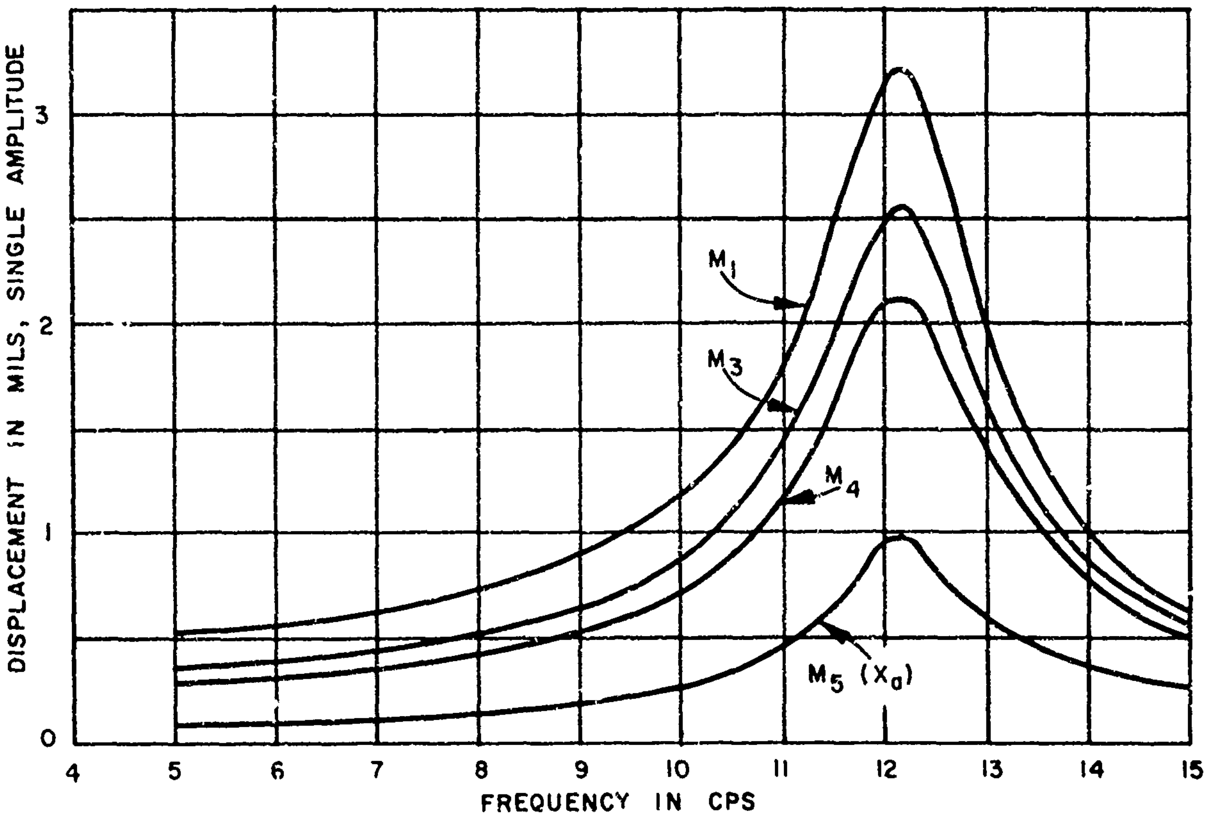

Figure 17 - Computed Response of System Shown in Figure 16 with C_p = 4100 and an Exciting Force of ±1000 Pounds

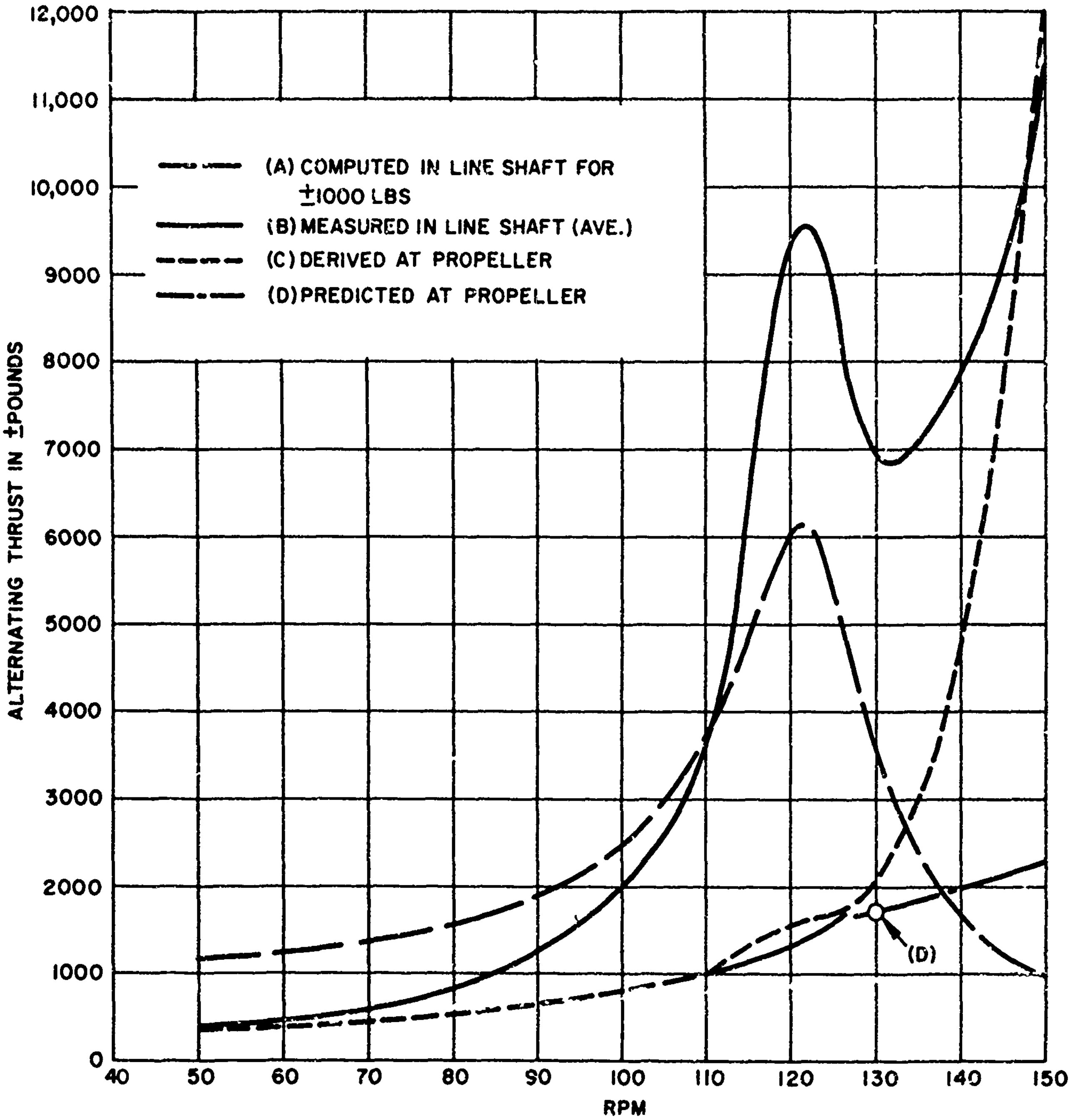

Figure 18 - Alternating Thrust at Propeller and in Line Shaft

second mode is approached, raising doubt as to the validity of the calculations in that area. However, the mass-elastic system used here was determined considering the actual second mode frequency and should be fairly accurate. at least through the operating range.

The estimate of the exciting force used by Zaloumis (see footnote on page 2) in his response prediction was based on a wake survey at 130 rpm, using the open-water propeller characteristics to determine the thrust in relation to the J-values. This preliminary calculation considered only the axial velocity components at 0.7 times the radius and yielded an alternating thrust of ±12,600 lb. On this basis, and the assumption that the alternating thrust varies as the square of revolutions per minute, response calculations were made. A more complete calculation of alternating thrust was made using the Burrill method, considering axial and tengential velocity components as well as longitudinal components over the entire blade.* This calculation resulted in a much smaller alternating thrust of ±1,700 lb at 130 rpm, which is close to the average alternating thrust as derived in this report and shown in Figure 18.

If the assumption is made that alternating thrust varies as the square of rpm, as is often done, the resulting calculated alternating thrust (curve (d) of Figure 18) is close to the derived curve below 130 rpm. Above 130 rpm, agreement is obviously poor.

Damping

The damping can be found from the relationship[2]

$$x_p = \frac{F}{\omega C_p} \text{ or } C_p = \frac{F}{\omega X_p}$$

---

*Recorded in a Taylor Model Basin letter report now being prepared by W. H. Hinterthan titled "Propeller Excited Vibratory Forces for Submarine Tender AS-33, USS SIMON LAKE."

where x_p is amplitude of propeller,

F is amplitude of exciting force,

ω is circular frequency, and

C_p is equivalent viscous damping at the propeller.

This relationship is true for a system with propeller damping only and only at resonance where the input energy is equal to the damping energy. At resonance it was found that F = ±1400 lb, and x_p = 0.005 in. Therefore:

$$C_p = \frac{1400}{2\pi \times 12.2 \times 0.005} = 3{,}660 \frac{\text{lb-sec}}{\text{in}}$$

The response curves (Figure 19) of the equivalent system which has been derived are obtained by adjusting the curves in Figure 17 to account for the derived exciting force instead of a constant ±1000 lb exciting force and adjusting the amplitudes at resonance to reflect the proper amount of damping. The magnification factor at the propeller (ratio of the dynamic propeller amplitude to the propeller amplitude when subjected to a static force of the same magnitude) at resonance is 7.5. The magnification factor of thrust at the thrust bearing is 7.9.

Superimposed on these curves are points which represent the measured response of the foundation mass at shaft level. The discrepancy above resonance is probably due to inaccuracies in the mass-elastic system which yield the correct natural frequencies but apparently not the correct mode shape when the second mode is approached.

CONCLUSIONS

1. A longitudinal resonance exists in the operating range of the shaft at about 81 percent of full power rpm.

2. The severity of this resonance is of no consequence since the amplitudes of vibration are low, in fact, considerably lower than usual.

3. The major reason for these low measured amplitudes is that the exciting forces are only about 1 percent of the mean thrust, except at or near full power. In estimating the exciting forces from a wake survey, only 130 rpm was considered. Alternating thrust at that speed was calculated

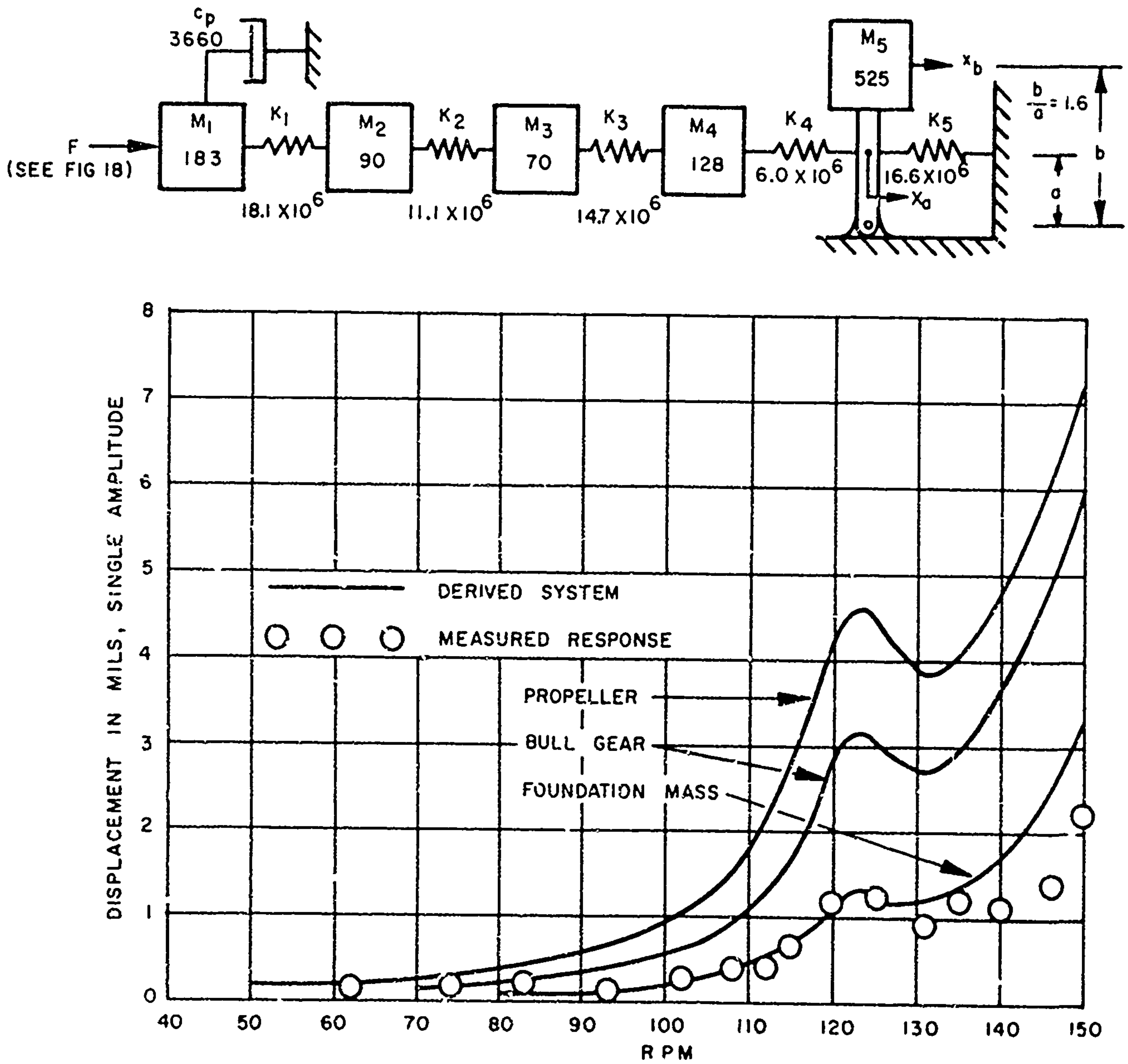

Figure 19 - Derived Equivalent System and its Amplitude Response to Derived Forces

to be ±1,700 lb. From that point the alternating thrust is normally assumed to vary as the square of rpm. Trial data indicate that:

A. The ±1,700 lb predicted is close to the average value (+2,000 lb) of actual alternating thrust at 130 rpm.

B. The alternating thrust on this ship does not vary as the square of rpm above 130 rpm. Since this result is contrary to an important assumption which is commonly accepted as fact, this point should be investigated on other ships.

4. Damping is only slightly more than expected, and falls between the Rigby method based on developed blade area and that based on blade edge length.

5. The condenser, turbines, gear case, and thrust-bearing housing vibrate more or less as one unit longitudinally--justifying the normal analytical procedure of lumping all of these into one foundation mass.

6. The condenser-turbine unit rotation due to longitudinal forces is negligible. This type of vibration was observed on ESSO GETTYSBURG,* which has an underslung condenser such as is shown in Figure 20. An underslung condenser typically results in the center of gravity of the turbine-condenser unit falling well below the level of turbine support. In designing SIMON LAKE, PSNS placed the after supports of the turbine-condenser unit on a line between the thrust bearing and the center of gravity of the turbine-condenser unit to eliminate the rocking vibration, on the assumption that the alternating-thrust forces will be transmitted approximately along this line. The trial indicated that the amplitudes of the turbine are slightly larger than those of the condenser. This might be due to the condenser-turbine mounting being too high. If it is assumed that forces from the thrust bearing are transmitted longitudinally through shear in the gear case, rather than diagonally from the thrust-bearing housing to the combined center of gravity of the condenser-turbine unit, then the mounts should be on the same level as the combined center of gravity instead of below it.

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*Information is taken from Taylor Model Basin letter report of 28 October 1964, titled "Preliminary Trip Report of First Underway Trial of SS ESSO-GETTYSBURG (Jul 1964).

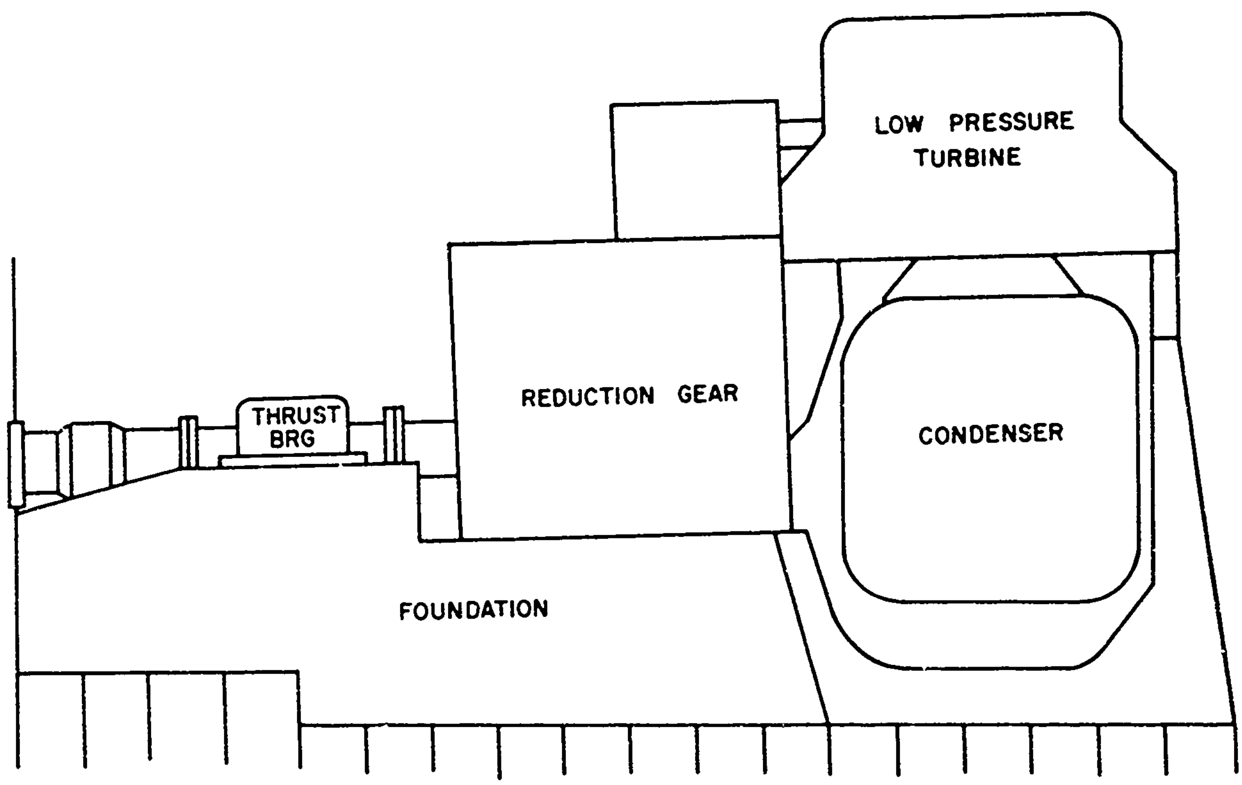

Figure 20 - Main Propulsion Plant of ESSO GETTYSBURG

7. Since there are two characteristics (long shafts and more blades on the propellers) of modern Navy ships which bring the first longitudinal mode and at least the approach to the second mode resonance into the operating range, a special effort must be made to improve our methods of predicting longitudinal shaft vibration above the first mode. The discussion of the "lever effect" in this report is an effort in this direction.

8. Reference 3 requires that Navy ships have no longitudinal critical speeds from 50 to 115 percent of full power rpm. The fact that SIMON LAKE has a critical speed in the middle of this range that is not detrimental indicates that this requirement is not always a realistic one. At present the Model Basin is preparing recommendations for limited changes to Military Standard 167 and, at the same time, conducting a program of research which will provide a basis for a more complete revision later.

RECOMMENDATIONS

1. It is recommended that the Model Basin conduct underway trials on a propulsion system which has the turbine-condenser mounts at or near the level of the center of gravity of the turbine-condenser unit. A moderate amount of research in this area may make it possible to specify the optimum position of mounts for use in future design.

2. In future model-wake surveys, it is recommended that the Model Basin evaluate the alternating torque and thrust at three speeds--60, 85, and 100 percent of full power. It is also recommended that full-scale measurements be made for comparison. In these studies, the relationship between the average values and the peak values of alternating forces should be studied.

3. It is recommended that future trials of longitudinal shaft vibration be designed to investigate, as much as is possible, the second mode of vibration, and that analytical studies be conducted in conjunction with these in an effort to develop a prediction technique which is accurate for the second mode as well as the first. This will include, primarily, a better definition of the foundation mass and spring constant.

ACKNOWLEDGMENTS

The officers and crew of SIMON LAKE were most cooperative while conducting the trials. The author is indebted to Mr. W. Huber for recording the data, to Mr. A. Zaloumis for his guidance throughout the preparation of this report, and to Mr. W. Hinterthan for his estimates of alternating thrust from the wake survey.

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13 ABSTRACT

Measurements were made on the propulsion system of the USS SIMON LAKE (AS-33) in February 1965 as part of a program to improve analysis procedures used by the Navy for predicting the longitudinal vibration of shaft propulsion systems. The objectives were to find the axial exciting forces and damping associated with the propulsion system of this ship, as well as to determine how the gear case, turbines, condenser, and machinery foundation affect longitudinal vibration. Alternating thrust in the shaft and longitudinal displacement of the gear case, low-pressure turbine, condenser, and machinery foundation were measured. A resonance was found to exist in the operating range, but it is not considered detrimental. The gear case, turbines, and condenser move essentially as one unit. A mass-elastic system derived from measured data includes a lever effect acting on the foundation mass. The exciting forces are lower than usual, except at or near full power.

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